(12) INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(19) World Intellectual Property Organization
International Bureau

(43) International Publication Date
17 July 2003 (17.07.2003)

PCT

(10) International Publication Number
WO 03/058034 A1

(51) International Patent Classification[7]: F01B 3/00, F04B 1/20

(21) International Application Number: PCT/NL03/00015

(22) International Filing Date: 10 January 2003 (10.01.2003)

(25) Filing Language: Dutch

(26) Publication Language: English

(30) Priority Data:
1019736 12 January 2002 (12.01.2002) NL
1020932 24 June 2002 (24.06.2002) NL

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(81) Designated States *(national)*: AE, AG, AL, AU, BA, BB, BR, BZ, CA, CN, CO, CR, CU, DM, DZ, EC, GD, GE, HR, HU, ID, IL, IN, IS, JP, KE, KP, KR, LC, LK, LR, LT, LV, MA, MG, MK, MN, MX, NO, NZ, OM, PH, PL, RO, SC, SG, TN, TT, UA, US, UZ, VC, VN, YU, ZA.

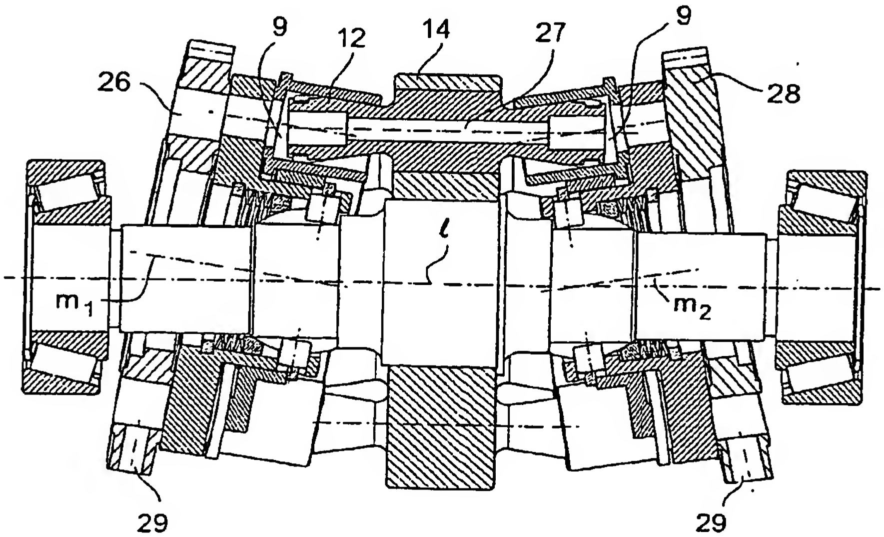
(84) Designated States *(regional)*: ARIPO patent (GH, GM, KE, LS, MW, MZ, SD, SL, SZ, TZ, UG, ZM, ZW), Eurasian patent (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European patent (AT, BE, BG, CH, CY, CZ, DE, DK, EE, ES, FI, FR, GB, GR, HU, IE, IT, LU, MC, NL, PT, SE, SI, SK, TR), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN, GQ, GW, ML, MR, NE, SN, TD, TG).

Declaration under Rule 4.17:
— *as to the applicant's entitlement to claim the priority of the earlier application (Rule 4.17(iii)) for all designations*

Published:
— *with international search report*

[Continued on next page]

(54) Title: HYDRAULIC DEVICE



(57) Abstract: The invention relates to a hydraulic device having, in a housing, a rotor, which can rotate about a first axis, with pistons, chambers, which can rotate about a second axis and are formed by a cylindrical wall and a piston. The chamber is connected to a line connection via a first passage, of which a face-plate port forms part. On the side remote from the first face plate, the rotor is provided with rotor ports and with a second passage through each piston for connecting each chamber to a rotor port, it being possible for the rotor ports to rotate along the housing or along a second face plate, which is positioned in the housing, in such a manner as to form a seal. As a result, the rotor can be balanced in the axial direction, so that the axial load is independent of the pressure in the chambers.

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Hydraulic device

The invention relates to a device according to the preamble of Claim 1. Devices of this type are known. The drawback of the known device is that an axial force which is dependent on the pressure in the chambers is exerted on the rotor. As a result, the bearing arrangement of the rotor is subject to fluctuating loads, and deformation of the bearings is dependent on the pressure and is greater at higher pressure. The result of this is that at higher pressures larger leakage gaps are formed, with the result that additional leakage occurs in particular at higher pressures, which constitutes a drawback.

To avoid this drawback, the hydraulic device is designed in accordance with the characterizing part of Claim 1. The result of this is that the oil pressure is active on two sides of the rotor and the axial force can be balanced. The load on the bearings is then more or less independent of the pressure in the chambers, and leakage is minimized.

According to a further improvement, the device is designed in accordance with Claim 2. The result of this is that it is possible to supply and discharge the oil to and from the chambers via two different face plates. In this case, it is possible for a face-plate port at one face place to be designed to be closed over part of its circumference, so that it closes off an opening in the housing. As a result, it is possible to rotate the face plate over a greater arc length than the arc length of the face-plate port, and the control range of the device is increased in a simple way through rotation of the face plate.

According to a further improvement, the device is designed in accordance with Claim 3. The result of this

is that the pressure peaks which occur when the drum plates are being closed by the face plate are limited, since for each chamber the oil stream can flow along two face plates. This improves the efficiency and reduces the noise levels produced.

The invention also comprises a device in accordance with the preamble of Claim 4. Devices of this type are known. The drawback of the known device is that to increase the displacement the diameter of the rotating parts has to be increased, with the result that the manufacturing tolerances and the leakage gaps are also enlarged.

To avoid this drawback, the device is designed in accordance with the characterizing part of Claim 4. The result of this is that the displacement per revolution is doubled in a simple way by doubling the number of pistons, while the surface area of the face-plate ports is also doubled, so that the losses do not increase.

According to a further improvement, the device is designed in accordance with Claim 5. The result of this is that pistons on either side of the rotor alternately move past the face-plate ports, so that it is possible to count on the total number of pistons when considering the pulses in the flow of oil and torque during rotation of the rotor. Consequently, the magnitude of these pulses is reduced.

According to a further improvement, the device is designed in accordance with Claim 6. When three or more face-plate ports are used, the hydraulic device can be used as a hydraulic transformer, in which chambers are closed off by the face-plate ports while the volume in the chambers is changing greatly. If the number of pistons amounts to a multiple of the number of face-plate ports, the axial force acting on the drum plate

remains more or less constant, with the result that it can rotate more smoothly and stably.

According to a further improvement, the device is designed in accordance with Claim 7. The result of this is that the opening and closing of corresponding chambers on both sides of the rotor does not take place at the same rotational position, with the result that torque fluctuations and pressure peaks in a chamber can be avoided. As a result, the stability and efficiency are improved.

According to one improvement, the device is designed in accordance with Claim 8. The result of this is that the excessive noise pollution resulting from pressure pulses which may arise as a result of resonance in the connection passages is to a significant extent suppressed, reduced or prevented.

According to one improvement, the device is designed in accordance with Claim 9. The result of this is that pressure pulses which may occur as a result of resonance in the connection passages are suppressed, reduced or prevented by simple means.

The invention is explained below with reference to a number of exemplary embodiments and with the aid of a drawing, in which:

Figure 1 shows a cross section through the interior of a hydraulic device,

Figure 2 shows a perspective view of the hydraulic device shown in Figure 1,

Figure 3 shows a detail from Figure 1 including the forces acting on the drum sleeve,

Figure 4 diagrammatically depicts the planes through the axes of the rotor and the drum plate,

Figure 5 shows a second embodiment of the hydraulic device,

Figure 6 shows a hydraulic device according to a third embodiment,
Figures 7 and 8 show a detail of an embodiment of the drum plate,
Figure 9 shows an embodiment of a drum sleeve for use in the hydraulic device,
Figure 10 shows a detail of the drum sleeve from Figure 9,
Figure 11 shows a first embodiment of internal securing of the drum sleeve to the drum plate,
Figure 12 shows a second embodiment of internal securing of the drum sleeve to the drum plate,
Figure 13 shows a first embodiment of a pump or motor, and
Figure 14 shows a second embodiment of a pump or motor.

The components shown in Figures 1 and 2 are the parts of a hydraulic transformer which are mounted in a housing. A hydraulic transformer of this type is described, for example, in the published applications WO 9731185 and WO 9940318, the contents of which are deemed to be known. Bearings 1 in which a rotor shaft 2 having an axis l can rotate are mounted in the housing in a known way. A rotor 14 with rotor holes 15 is mounted on the rotor shaft 2. In the rotor holes 15 there are rod-like components which form pistons 12 on either side of the rotor 14. The pistons 12 are provided with piston rings 10, the outer surface of the piston rings 10 being convex in shape, and the centre of this convexity lying in a single plane for all the pistons on one side of the rotor 14. If appropriate, the outer surface of the piston rings 10 is arched. The left-hand side and the right-hand side of the rotor 14 are symmetrical with respect to the centre of the rotor 14. Each side of the rotor 14 interacts with a drum plate 7 with drum sleeves 11 which rotate about an axis m_1 and m_2, the axes l and m_1 and l and m_2, respectively, intersecting one another in the plane perpendicular to l through the centre points of the outer surfaces of

the piston rings 10 for the pistons 12 located on that side.

On the rotor shaft 2 there is a centring surface 22 about which the drum plate 7 can pivot. The centring surface 22 is convex, the centre of the convexity lying in the plane on which the centre of the convex piston rings 10 lies. The rotation of the drum plate 7 is coupled to the rotation of rotor shaft 2 by means of a key 16 which engages in a keyway. In the plane of the surface of the shaft, the key 16 has a rounding radius which is smaller than the radius of the centring surface 22, so that the key 16 does not become jammed in the keyway when the drum plate 7 rotates. If appropriate, there may be more than one key 16. It is also possible for the key 16 to be mounted in the rotor shaft 2 and for the keyway to be arranged in the drum plate 7.

On the side which faces the pistons 12, the drum plate 7 is provided with drum sleeves 11 which are clamped against the drum plate 7 by a sleeve holder 18. On the inner side, the drum sleeve 11 has a cylindrical wall 23. Each piston 12 is surrounded by a drum sleeve 11, it being possible for the piston ring 10 to move in a sealed manner along the cylindrical wall 23. The piston 12 and the cylindrical sleeve 11 therefore form a chamber 9, the volume of which changes when the rotor shaft 2 rotates. The change in volume causes oil flow into and out of the chamber 9 via a drum sleeve opening 24, a drum port 6 and a drum-plate port 3 to an opening in the housing. The corresponding drum-plate ports 3 are connected to one another in the housing. Since the axes of rotation of the rotor 14 and the drum plate 7 form an angle with respect to one another, the pistons 12 in the plane of the drum plate 7 describe an elliptical path, and the drum sleeves 11 will slide over a contact surface 8 of the drum plate 7. The holder 18 is designed with openings which allow this

sliding to take place, and it also ensures that the gap between drum plate 7 and drum sleeve 11 remains limited, so that pressure can build up in the chamber 9 when starting up. In another embodiment, it is also possible for the holder 18 to be secured in such a manner to the drum plate 7 that the rotation of the rotor 14 is transmitted via the pistons 12, the drum sleeves 11 and the holder 18 to the drum plate 7, with the result that the key 16 and the associated keyway can be dispensed with.

The face-plate port 3 is arranged in a face plate 4 which is supported against a surface of the housing. This surface is not at right angles to the axis 1, but rather forms and angle therewith, thus determining the direction of the axis m_1 or m_2 and therefore also the rotational position at which the volume in the chamber 9 is at its minimum or maximum. The face plate 4 is secured in the housing in such a manner that it can rotate about the axis m_1 or m_2 and is provided over part of its circumference with toothing 5 which interacts with a pinion driven by a drive. A centring sleeve (not shown) can be used to centre the rotation of the face plate 4 in the housing in a known way. Rotation of the face plate 4 causes the setting of the hydraulic transformer to change, as described in the patent applications which were cited earlier in the text.

To keep the openings between face plate 4 and drum plate 7 small during starting up, when there is as yet no pressure in the chambers 9, there is a pressure-exerting ring 19 which is supported against the centring surface 22. Between the pressure-exerting ring 19 and a ring 21 secured in the drum plate 7 there are cup springs 20, by means of which the drum plate 7 is always pressed onto the face plate 4. If appropriate, other resilient elements may be used instead of cup springs 20.

Figure 3 shows the drum sleeve 11, which is supported on the contact surface 8 of the drum plate 7. During use, a high pressure prevails in the chamber 9 and the drum port 6, while a lower pressure prevails outside the drum sleeve 11. A changing oil pressure will form in the gap in the contact surface 8 between drum sleeve 11 and drum plate 7, as indicated by arrows A in the figure. To prevent the size of the gap from increasing under the influence of this oil pressure, the drum-sleeve opening 24 has a smaller surface area than the sealing surface of the piston 12 in the cylindrical wall 23. There is now a rim around the drum-sleeve opening 24, on which oil pressure, indicated by arrows B, exerts a force on the drum sleeve 11 in the direction of the contact surface 8. If the drum sleeve 11 is dimensioned correctly, it is possible to ensure that under the influence of the oil pressure the drum sleeves 11 are always pressed onto the contact surface 8.

The forces acting on the piston ring 10 are also shown in Figure 3. On the outer side, the piston ring 10 has a convex surface, so that the seal between piston ring 10 and the cylindrical surface 23 is produced in the plane which is perpendicular to the cylindrical surface 23, i.e. perpendicular to the axis m. If appropriate, the surface may be arched rather than circularly convex. The piston ring 10 is not subject to uniform load all the way around as a result of the angles between the axes l and m, since the surface area which is under high pressure on the outer side as a result of oil is large at E, as indicated by arrows, and is small at D. Since the surface area which is under pressure is small at D, the piston ring 10, under the influence of the pressure on the inner side, which is indicated by the arrows C, could press heavily on the cylindrical wall 23 and cause a high frictional force.

This frictional force is greatly reduced through the fact that the inner side of the piston ring 10 is designed with a shoulder 25. If this shoulder 25 lies halfway along the width of the piston ring 10, the outwardly directed force is halved. As shown, the inwardly directed force at E is greater than the outwardly directed force. Under the influence of this, the piston ring 10 is supported on the piston 12, while as a result of the displacement of the drum sleeve 11 the seal between piston ring 10 and cylindrical wall 23 is retained all the way around. As a result of the support, the piston ring 10 exerts a resulting force R on the piston 12, and this force R drives the rotor 14. Obviously, it is also possible for the device to be fitted without piston rings 10, but in this case it will be necessary to take measures to avoid contamination which may cause wear.

The hydraulic transformer is designed in such a manner that the pistons 12 on either side of the rotor 14 alternately move into the top dead centre, i.e. the position where the volume of the chambers 9 is at its minimum, so that in terms of fluctuations in the oil flow and the torque acting on the rotor 14, it is possible to count on the total number of pistons 12, i.e. eighteen pistons 12 in the example shown. In the exemplary embodiment shown, in which the pistons 12 on either side of the rotor 14 lie in line with one another, this is achieved by rotating the top dead centre of the pistons on one side through an angle α with respect to the top dead centre on the other side. In this case, α is equal to half the rotational angle between two pistons 12. The face plates 4 are also rotated through this angle with respect to one another. This is shown in Figure 4a, in which V_1 is the plane through the axes l and m_1, and V_2 is the plane through the axes l and m_2. Another embodiment is shown in Figure 4b. In this case, the axes l, m_1 and m_2 lie in a plane V and the pistons 12 are arranged offset in the

rotor 14. This embodiment is of interest in particular if the volumes of the chambers 9 which successively acquire a maximum volume are coupled through passages with valves as discussed in applications WO 0244524 and WO 0244525. In the embodiment shown in Figure 4b, axes of the pistons 12 are parallel to the axis 1, and the pistons on either side are different components which are arranged offset in the rotor 14. In an embodiment which is not shown and in which the pistons 12 on either side of the rotor 14 are offset and the axes 1, m_1 and m_2 likewise lie in one plane, the pistons 12 on either side are made from a component which is mounted in the rotor 14 and has an axis which forms an angle with the axis 1.

It is preferable for the rotation of the two face plates 4 to be coupled, so that only one drive is required. This is achieved, for example, by rotating the face plates 4 using a gearwheel, coupled to a shaft and coupling the two shafts to a homokinetic coupling, so that the rotation of the two face plates is accurately synchronous. If appropriate, the two face plates 4 may be provided with their own drive, so that for certain operating states a hydraulic preloading can be obtained.

The angle β between the axes 1 and m determines the displacement of the device. In the embodiment shown, with 9 pistons 12 on each side, the angle is 9 degrees. If the number of pistons 12 increases, this angle has to be smaller, since otherwise the constriction of piston 12 which is required in order always to remain clear of the drum sleeve 11 becomes too great. In the embodiment shown, calculations have been based on a maximum rotational speed of the rotor 14 of 8000 revolutions per minute. If this speed is greater, a smaller angle β is required in order to prevent the occurrence of unacceptable pressure peaks.

In the exemplary embodiment shown, it is shown that the drum plate 7 is centred by means of the centring surface 22. It is also possible for this centring to be designed in other ways, for example by providing the drum plate 7 with a spherical bearing on its outer circumference, which is secured in the housing. Another embodiment may involve the drum plate 7 being centred with respect to the face plate 4, for example by providing the latter with a conical shape. It is also possible for a centring sleeve to be positioned in the housing in order to centre both the face plate 4 and the drum plate 7.

Figure 5 shows another embodiment of the hydraulic transformer. In this case, the axes l, m_1 and m_2 of the rotor 14 and both drums may lie in a single plane, although it is also possible for them to be designed as shown in Figure 4a. The chambers 9 on either side of the rotor 14 are connected to one another by a passage 27 running through the pistons 12. Face plates 26 and 28 are designed in such a manner that the face-plate port 3 leading to the tank connection is directly connected to the interior of the housing via a passage 29, this interior being connected to the tank connection. The face plates 26 and 28 are designed in such a manner that of the remaining two face-plate ports 3, each face plate 26 or 28 has one of the two ports and is closed at the location of the other port. This makes it possible for the connection in the housing to have an opening to the face plate over a wide angle and enables the face plates to rotate through a large angle, with the result that the control range of the hydraulic transformer is increased in a simple manner through rotation of the face plate. The rotation of the face plates 26 and 28 is coupled in the manner described above.

In the exemplary embodiments given above, the device has been described as a hydraulic transformer. It will be clear to the person skilled in the art that the

device can be made suitable for use as a pump or a motor with only minor adjustments, such as, inter alia, to the face plates 4 and the rotor shaft 2. Examples of this are shown in Figures 13 and 14, which will be discussed later on in the text.

Figure 6 shows an exemplary embodiment in which pistons 12 are accommodated on only one side. Their design corresponds to that which has been described in the embodiment shown in Figures 1 and 2. For axial balancing of the rotor 14, the latter is provided, on the side remote from the piston, with a face plate 34. On the side of the face plate 34, the rotor 14 is provided with chambers 31 which, via a passage 30, are in communication with the chambers 9. The surface area of the chambers 31 is comparable to the sealing surface area of the pistons 12, so that the rotor 14 is balanced in the axial direction.

The face plate 34 may be designed without face-plate ports. In one embodiment, there may also be face-plate ports 33, which are in communication with passages in the housing. This makes it possible to reduce pulses in the liquid flow and liquid pressure, because the flow of liquid to and from the chamber 9 take place via two face plates.

In the exemplary embodiment shown in Figure 6, the rotor shaft 2 has been lengthened to outside the housing and ends at a shaft end 37. The rotor shaft 2 is for this purpose provided with a seal 36 and a bearing 35. This embodiment is particularly suitable for use as a pump or motor.

In the exemplary embodiments discussed above, the angles between the axes are constant and the displacement is varied through rotation of the face plates. Obviously, the design of the rotor with the fixedly mounted pistons and the drum plate with the

drum sleeves which can be displaced perpendicular to the axis of the drum plate can also be used in embodiments in which the axis of the drum plate can pivot with respect to the axis of the rotor.

Figures 7 and 8 show a modified embodiment of the drum plate 7 which simplifies the sliding of the drum sleeves 11 over the contact surface 8. To reduce the resistance during the sliding movement of the drum sleeves 11 over the drum plate 7, it is necessary for a film of oil to be present between the drum sleeve 11 and the drum plate 7, even when the rotor 14 is stationary, so that the starting of the rotation of the rotor 14 is impeded to the minimum possible extent. To promote the formation of a film of oil of this type, the contact surface 8 has a curvature in one direction, so that there is linear contact between the drum sleeves 11 and the drum plate. For this purpose, the contact surface 8 is preferably designed as a cone with an angle 40 of 0.3 degree with a tolerance of ± 0.1 degree. The drum sleeve 11 now rests against a curved surface with a radius R_1 on the internal diameter of the drum plate and a radius R_2 on the outer side, R_2 being greater than R_1. Under the influence of the pressure in the chamber and/or the rotation of the rotor 14, the drum sleeve 11 will to some extent roll along the contact surface 8, with a local gap of a few microns existing between the drum sleeve 11 and the contact surface 8. A film of oil will form in this gap, ensuring lubrication.

Figures 9 and 10 show an embodiment of the drum sleeve 11 in which the latter has been produced by chipless deformation. With this production method, the drum sleeves 11 can be produced accurately and at low cost from sheet material by, inter alia, forcing the sheet material over a mandrel until it reaches the desired shape and dimensions. In this case, an internal diameter D_1 is produced accurately, in such a manner

that after hardening of the sleeve the diameter has the desired value. The forcing operation results in the formation of a bottom surface 43 of the sleeve which has a flange 41. For bearing in a sealed manner against the contact surface 8, the bottom surface 43 is accurately remachined to form a sealing surface 47, for example by grinding. For the flange 41 to bear against the sleeve holder 18, it is if appropriate also ground, so that the flange 41 is at a fixed distance 42 from the sealing surface 47.

In the sealing surface 47, there is a groove 44 which, via a passage 46, is in communication with the outer circumference of the drum sleeve 11. This allows a film of oil to form between the drum sleeve 11 and the drum plate 7 as discussed in connection with Figure 3; in this embodiment, the diameter of the sealing surface 47 is larger than the diameter of the groove 44, so that the drum sleeve 11 has a larger surface area for support and tilting of the drum sleeve 11 is limited. If appropriate, a groove 45 with a smaller diameter than the groove 44 may be arranged in the sealing surface 47. As a result, the surface area over which ++the decreasing pressure between the drum sleeve 11 and the drum plate 7 is active is accurately defined.

In the embodiments of the drum sleeve 11 discussed above, the drum sleeve 11 is designed as a component made from one material. If appropriate, the drum sleeve 11 may be made form two materials which are joined to one another, in which case that part of the drum sleeve 11 which forms the sealing surface 47 is made from a bronze-containing material, in order to reduce the friction. This friction results from the rotation and sliding of the drum sleeve 11 with respect to the drum plate 7. In this case, the shape of the join between the two components of the drum sleeve 11 and the elasticity of the materials are selected in such a

manner that the join is closed up under the influence of the liquid pressure prevailing in the chamber 9.

Figures 11 and 12 show alternative embodiments of the clamping device for clamping the drum sleeves 11 against the drum plate 7. In the embodiment shown above, the drum sleeves 11 are surrounded by the sleeve holder 18 on the outer side. In the event of rapid rotation of the rotor 14, high centrifugal forces are applied to a drum sleeve 11. If the liquid pressure in the chamber 9 is low, the drum sleeve 11 is only pressed onto the drum plate 7 by a low force, and there is then a risk of elastic deformation to the sleeve holder 18 as a result of the centrifugal force, which may give rise to unacceptable leaks occurring between the drum plate 7 and the drum sleeve 11. If the drum sleeve 11 is positioned, in the manner shown in Figures 11 and 12, with a clamping sleeve 48 in the vicinity of the drum plates 7, this drawback is avoided. The internal diameter of the drum-sleeve opening 24 is dimensioned in such a manner that the drum sleeve 11 can slide around the clamping sleeve 48 over the drum plate 7 in order to follow the piston 12, the drum sleeve 11 being axially enclosed between a collar of the clamping sleeve 48 and the drum plate 7. Figures 11 and 12 show two examples of the way in which the clamping sleeve 48 is secured in the drum plate 7. In this context, it is important for the clamping sleeve 48 to be accurately positioned in the axial direction with respect to the drum plate 7. In this case, it is preferable for the clamping sleeve 48 to be secured in the drum port 6. In the embodiment shown in Figure 11, the clamping sleeve 48 is designed with resilient elements which clamp behind a rim in the drum port 6. In the embodiment shown in Figure 12, the clamping sleeve 48 is pressed onto a shoulder with a heavy press fit. In addition to the embodiments of the clamping sleeve 48 which are shown, it will be clear to the

person skilled in the art that the same technical effect can also be achieved with other embodiments.

Figure 13 shows a hydraulic pump or motor which is designed in a similar way to the hydraulic transformer which has been described with reference to Figures 1 - 4, and the corresponding components are provided with identical reference numerals. The pump or motor is composed of a housing 61 and a cover 55. Bearings 1 are mounted in the housing 61 and the cover 55, and the rotor shaft 2 can rotate with an axis of rotation 1 in the bearings 1. In the cover 55 there is an opening through which a shaft end 51 projects in order to couple the shaft 2 to a motor or a tool. There is a seal 53 arranged between the shaft end 51 and the cover 55. A rotor 14, in which the pistons 12 are arranged on either side, is positioned between the bearings 1 on the shaft 2. This pistons 12 move, in a manner which has already been discussed above in the drum sleeves 11 which are coupled to the drum plates 7. The drum plates 7 are coupled to the rotor shaft 2 and rotate with it, being supported against the face plates 4. The surface between the face plate 4 and the drum plate 7 is in this case not at right angles to the axis of rotation 1. The face plates 4 are mounted in the manner shown in Figure 4a and are provided at a lowest point with a locking hole 52 which interacts with a pin which is mounted in housing 61 or cover 55 and thereby determines the rotational position of the face plate 4.

There are two face-plate ports arranged in each face plate 4: a low-pressure port, which is connected via a connection passage 54 and a low-pressure line 59 to a low-pressure connection T, and a high-pressure port, which is connected via a connection passage 54 and a high-pressure line 62 to a high-pressure connection P. In the embodiment shown, the connection passages 54 are of approximately equal length before they meet at 60 and pass into the low-pressure line 59 or the high-

pressure line 62. The chambers 9 in the drum sleeves 11 on either side of the rotor 14 are alternately connected to the two converging connection passages 54, and therefore, in the event of unfavourable conditions, it is possible that the oil may start to resonate at 60, which can lead to pressure peaks and excessive noise in the low-pressure line 59 and/or the high-pressure line 62. There is also a risk of excessive noise when using hydraulic transformers with three pressure lines.

To limit this excessive noise, there are resonance dampers, as shown in Figure 13, if appropriate in each connection passage 54. Each resonance damper comprises a chamber 57 which is filled with oil and is connected, by means of a passage 56 of small cross section, to the connection passage 54. The oil-filled chamber 57 is formed by a cavity in a cover 58 which is secured in the housing 61 or the cover 55. The dimensions of the chamber 57 and the passage 56 are matched to the frequency of the pressure pulses which occur and the properties of the oil. Suitable selection of these parameters makes it possible, for example, to reduce the pulses in the high-pressure line 62 in a pump from 50 bar to approximately 1-3 bar.

Figure 14 shows a hydraulic pump or motor in which the length of the connection passages 54 leading to the face plates 4 differs on the two sides of the rotor 14. The pressure pulses are likewise limited in this way, albeit to a lesser extent, for example the pulses which occur in the pressure line 62 of a pump are reduced from 50 bar to pulses of 1-3 bar. However, this method has the advantage that the influence of the properties of the liquid is reduced. If appropriate, it is also possible for the resonance dampers as shown in Figure 13 also to be used in the connection passages 54 as shown in Figure 14.

The designs for reducing excessive noise in the case of a double hydraulic pump or motor may, of course, also be used where necessary to reduce the pulses which may arise in a double hydraulic transformer.

In the exemplary embodiments of the hydraulic device which have been discussed above, the figures have always shown a device with drum sleeves 11 which, during rotation, describe an elliptical path and pistons 12 which describe a circular path. It will be clear to the person skilled in the art that a number of the design details discussed can also be used in other known designs, such as designs in which the drum sleeves are assembled to form a drum and the pistons are arranged in such a manner that they can be pivoted or displaced into or onto a drum, or designs in which the drum sleeves 11 can move over the face plate 4 and a drum plate 7 is not used. Other designs which can also be combined with the exemplary embodiments described here are designed with a variable displacement, for example achieved by making the angle β variable.

Claims:

1. Hydraulic device, comprising a housing (55, 61) with line connections (59, 92) and, inside the housing, inter alia a rotor (14), which can rotate about a first axis (1) and has pistons (12), chambers (9), which can rotate about a second axis (m_1, m_2) and are formed, inter alia, by a cylindrical wall (23) and the piston (12), it being possible for the first axis (1) to form a first angle (β) with the second axis (m_1, m_2), and, between the chambers and the housing, a first face plate (4) with face-plate ports (3), it being possible for the face plate to form part of the housing, in such a manner that a face-plate port may be part of a first passage between a line connection and a chamber, characterized in that the rotor (14) is provided, on the side remote from the first face plate (4), with rotor ports (31) and with a second passage (30) through each piston (12) for connecting the rotor port to a chamber (9), and in which the rotor ports can rotate along the housing or a second face plate (34), which is positioned in the housing and may be part of the housing, in such a manner as to form a seal.

2. Hydraulic device according to Claim 1, in which the second face plate (34) has one or more face-plate ports (33).

3. Hydraulic device according to Claim 2, in which the first face plate (4) and the second face plate (34), when the rotor (14) is rotating, simultaneously open and close the first and second passages between chambers (9) and face-plate ports (26, 33).

4. Hydraulic device comprising a housing (55, 61) with line connections (59, 92) and, inside the housing, inter alia a rotor (14), which can rotate about a first axis (1) and has pistons (12), chambers (9), which can rotate about a second axis (m_1, m_2) and are formed,

inter alia, by a cylindrical wall (23) and the piston (12), it being possible for the first axis (l) to form a first angle (β) with the second axis (m_1, m_2), and, between the chambers and the housing, a first face plate (4) with face-plate ports (3), it being possible for the face plate to form part of the housing, in such a manner that a face-plate port may be part of a first passage between a line connection and a chamber, characterized in that the pistons (12) and first face plates (4) which interact therewith are arranged on both sides of the rotor (14) and the rotor (14) is provided with holes (15) in which there is a rod-shaped component which is a piston (12) on both sides of the rotor.

5. Hydraulic device according to Claim 4 in which planes (V_1, V_2) through the first axis (l) and the two second axes (m_1, m_2) form a second angle (α) with one another, where if the number of pistons on one side of the rotor is equal to n, the angle (α) is equal to (1+2k)*180°/n, where k is equal to 0 or an integer number.

6. Hydraulic device according to Claim 4 or 5, in which each first face plate (4) has three or more face-plate ports (3), and the number of pistons (12) which interact with a face plate is a multiple of the number of face-plate ports.

7. Hydraulic device according to one of Claims 4-6, in which the pistons (12) are each provided with a third passage (27) which connects the chambers (9) on either side of the rotor (14), and the face-plate ports (3) of both first face plates (4) are designed identically in mirror-symmetrical fashion, and both first face plates are mounted in such a manner that the first passages open and close when the rotor is in different rotational positions.

8. Hydraulic device according to one of Claims 4-7, in which corresponding face-plate ports (3) are connected, by means of a connection passage (54), to a common line (59, 62), and in which a connection passage is connected through a damping passage (56) to a resonance chamber (57).

9. Hydraulic device according to one of Claims 4-8, in which corresponding face-plate ports (3) are connected, by means of a connection passage (54), to a common line (59, 62), and in which the length of the connection passages differs.

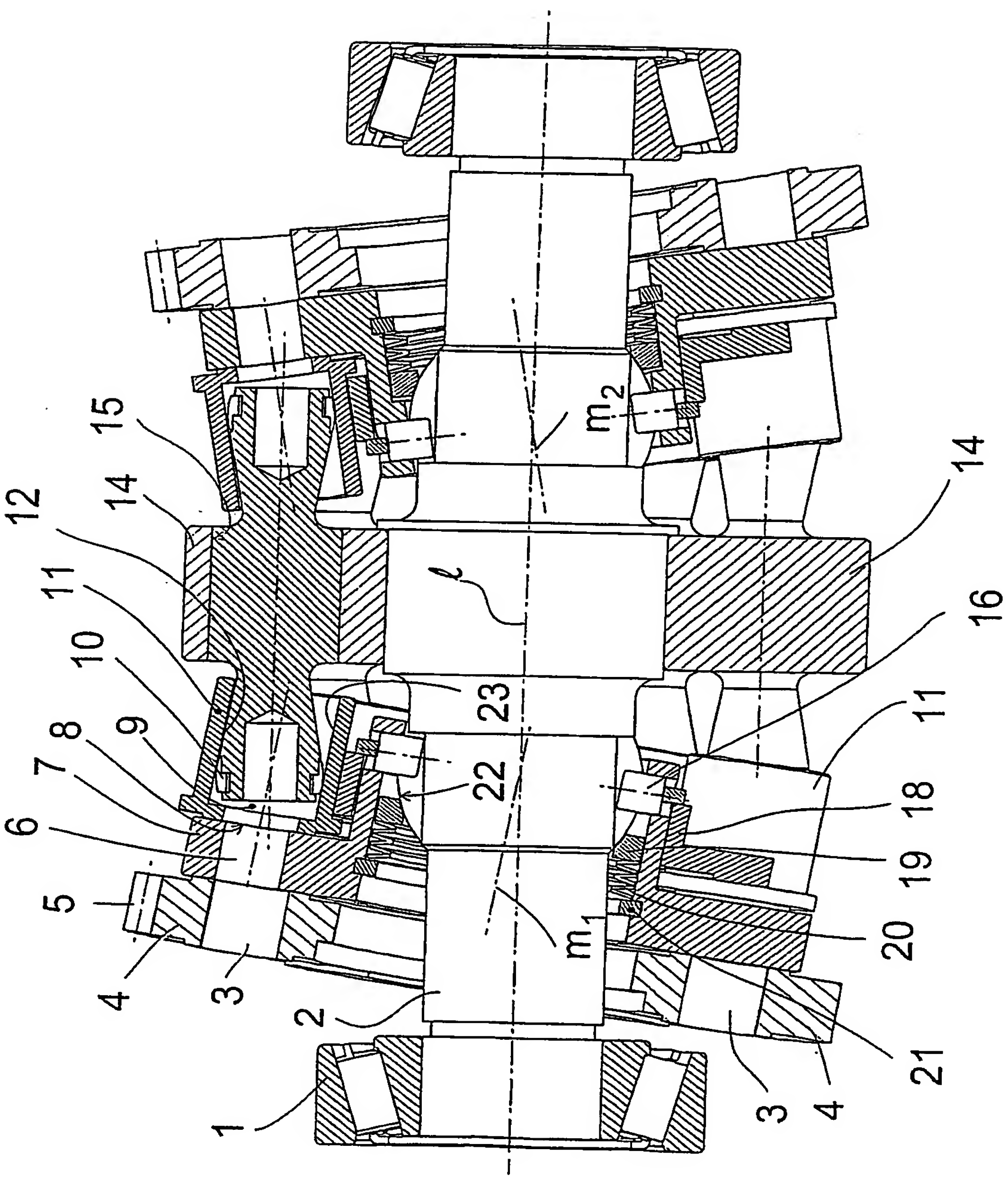

Fig. 1

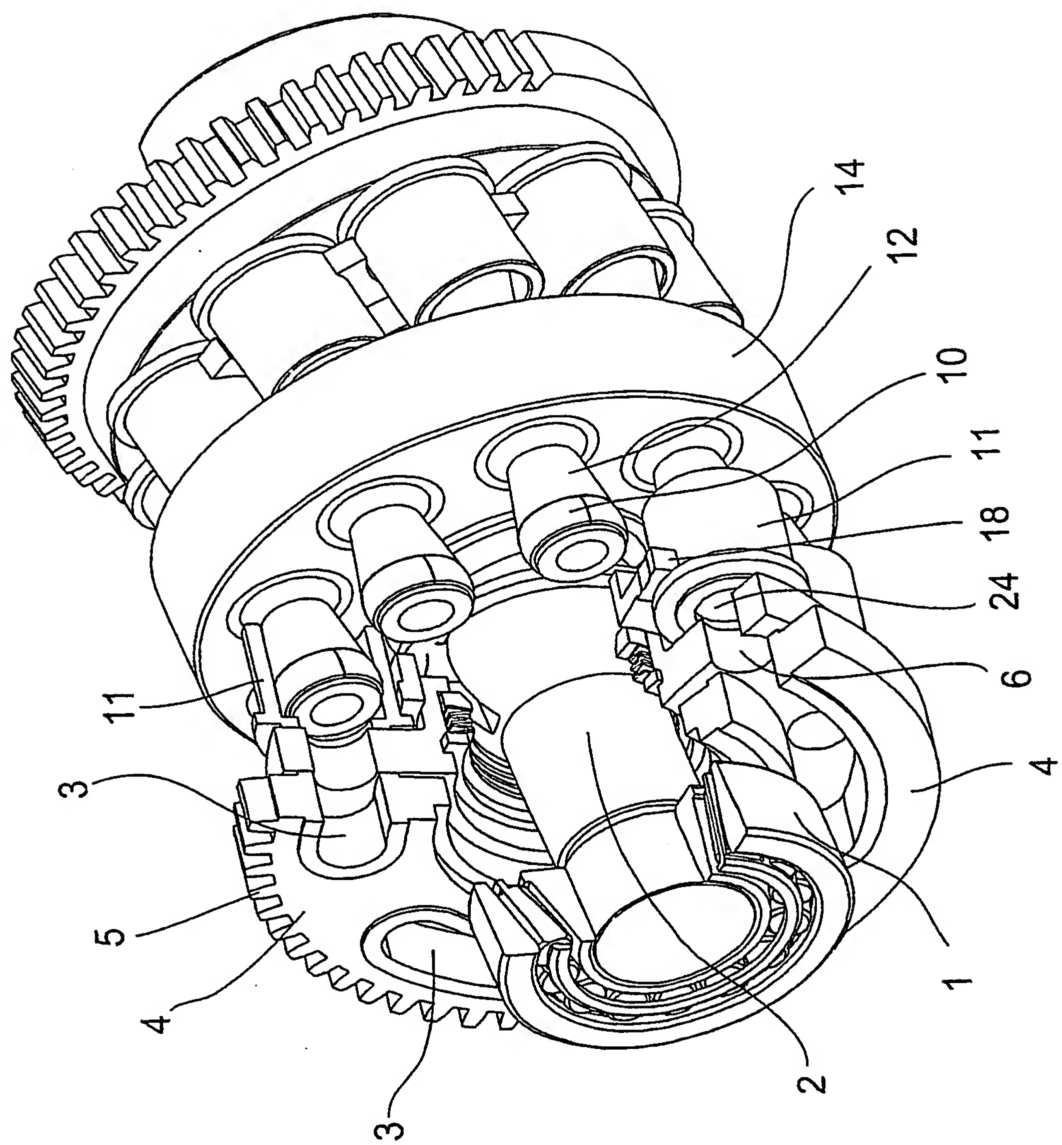

Fig. 2

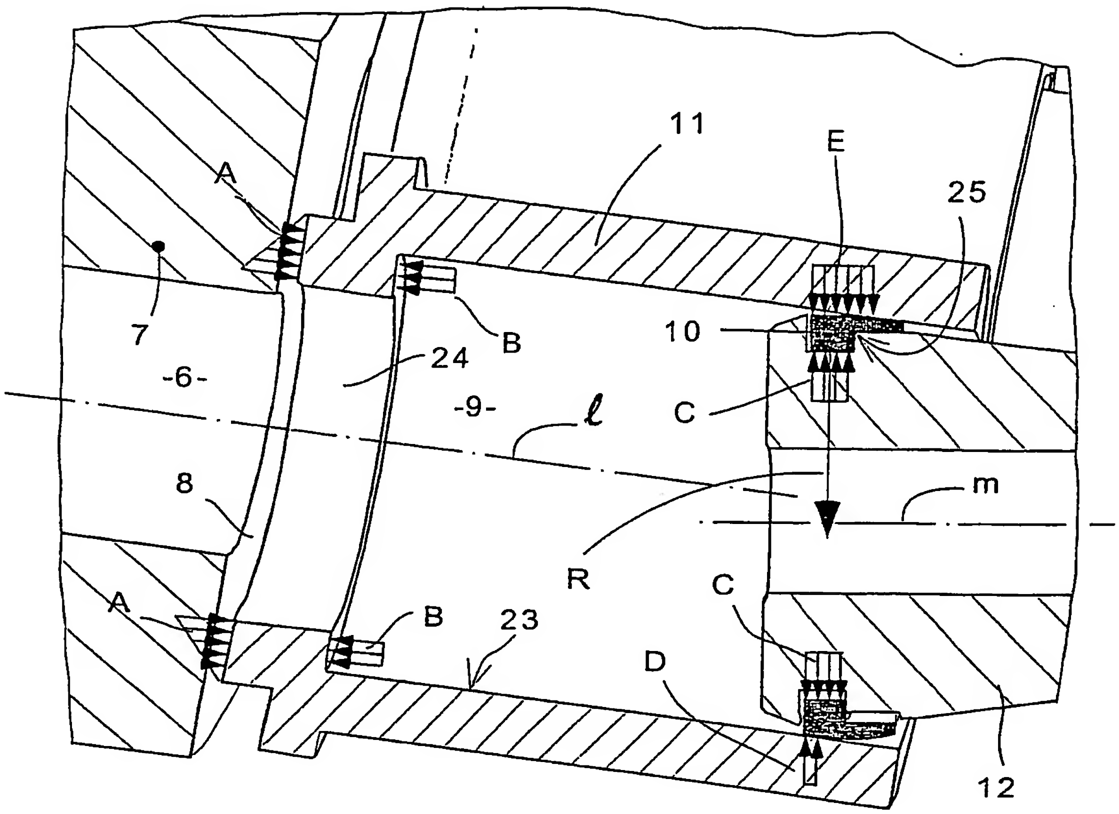

Fig. 3

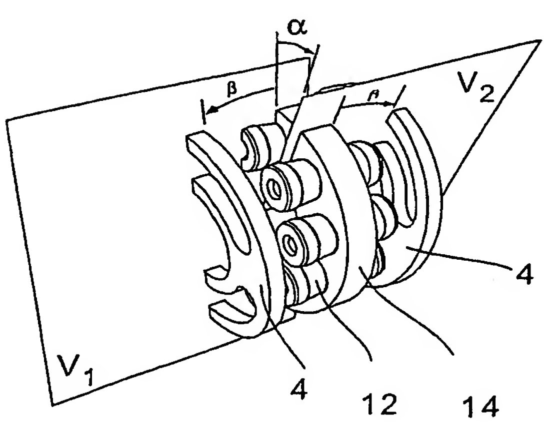

Fig 4a

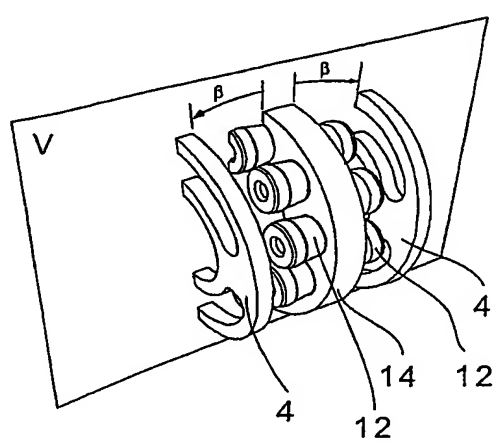

Fig 4b

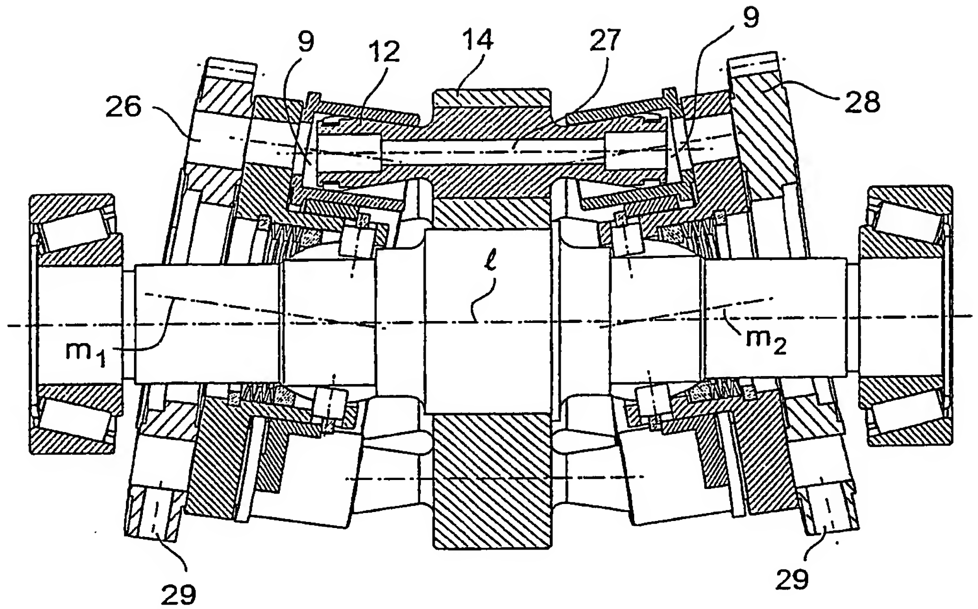

Fig 5

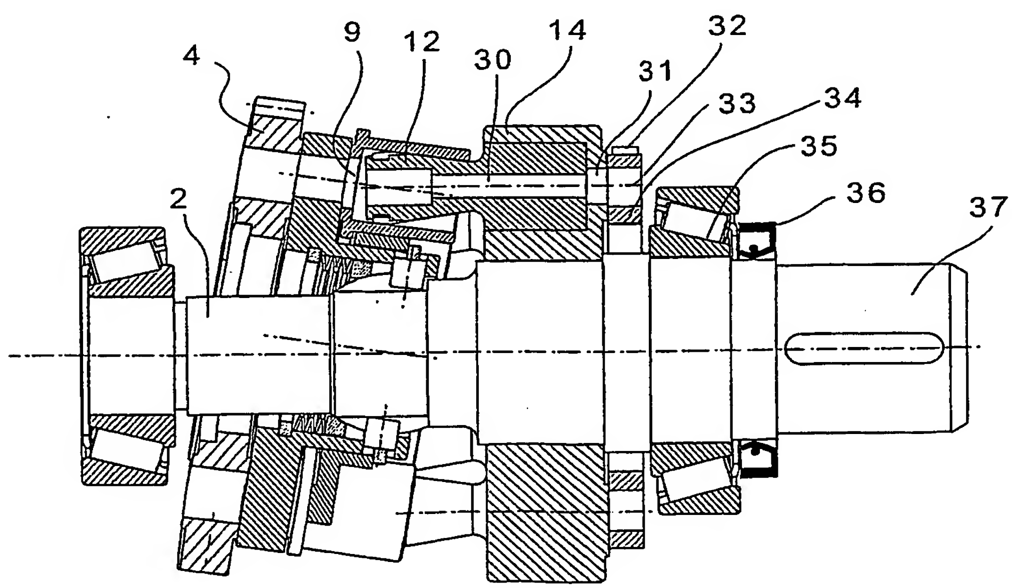

Fig 6

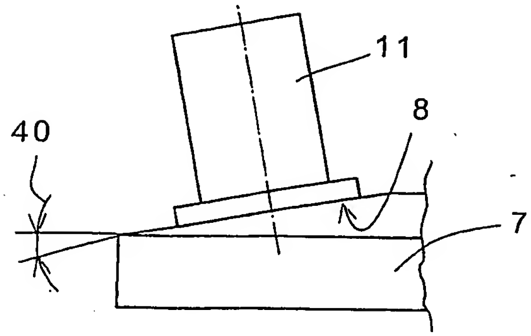

Fig. 7

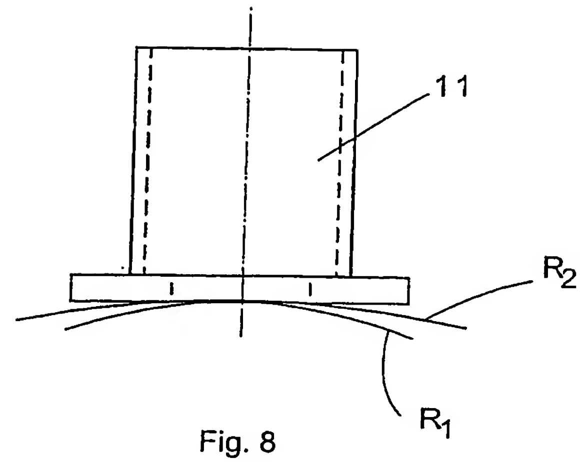

Fig. 8

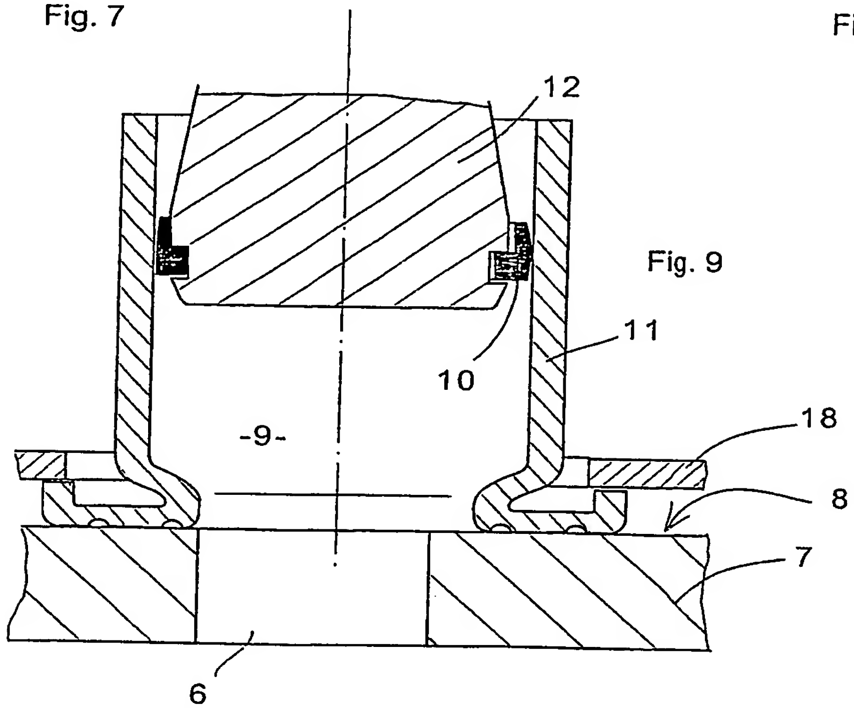

Fig. 9

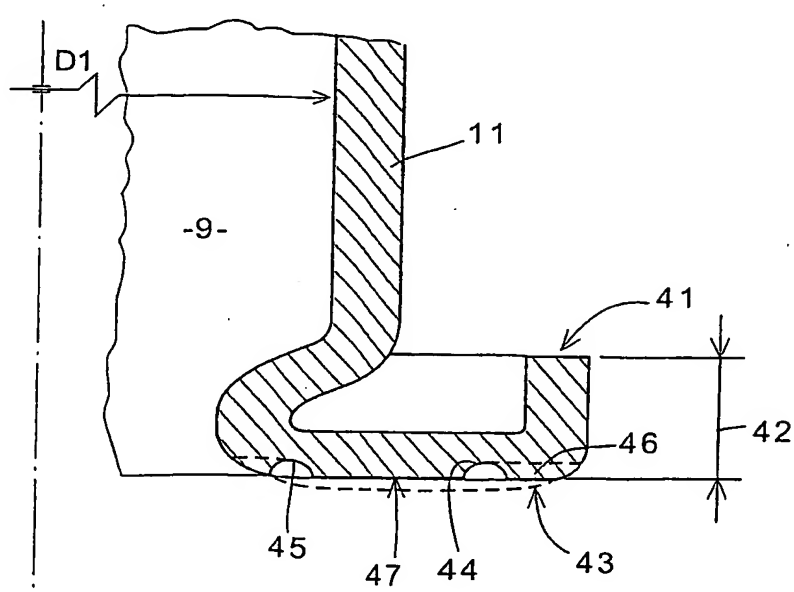

Fig. 10

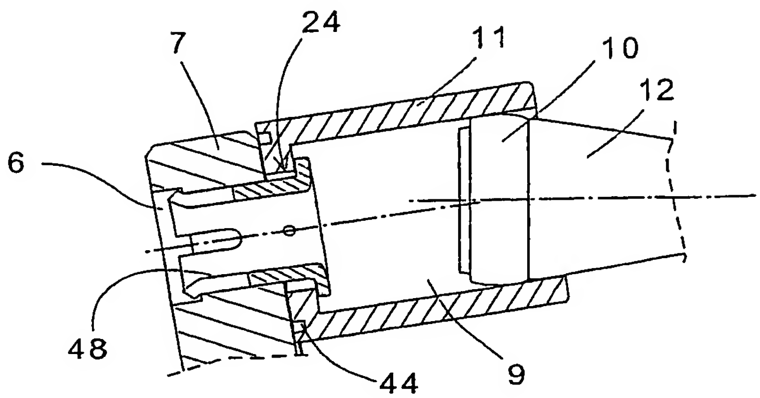

Fig. 11

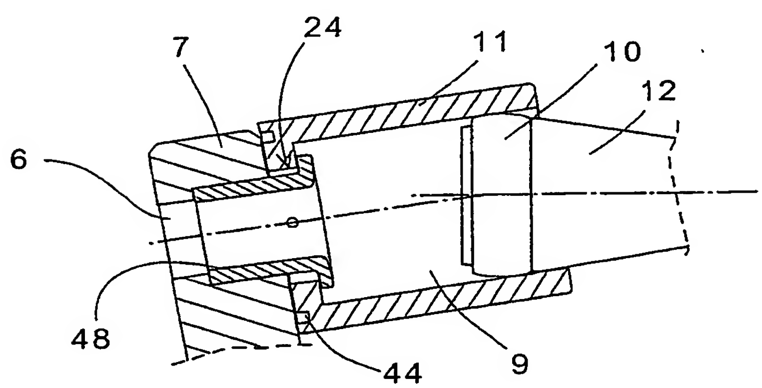

Fig. 12

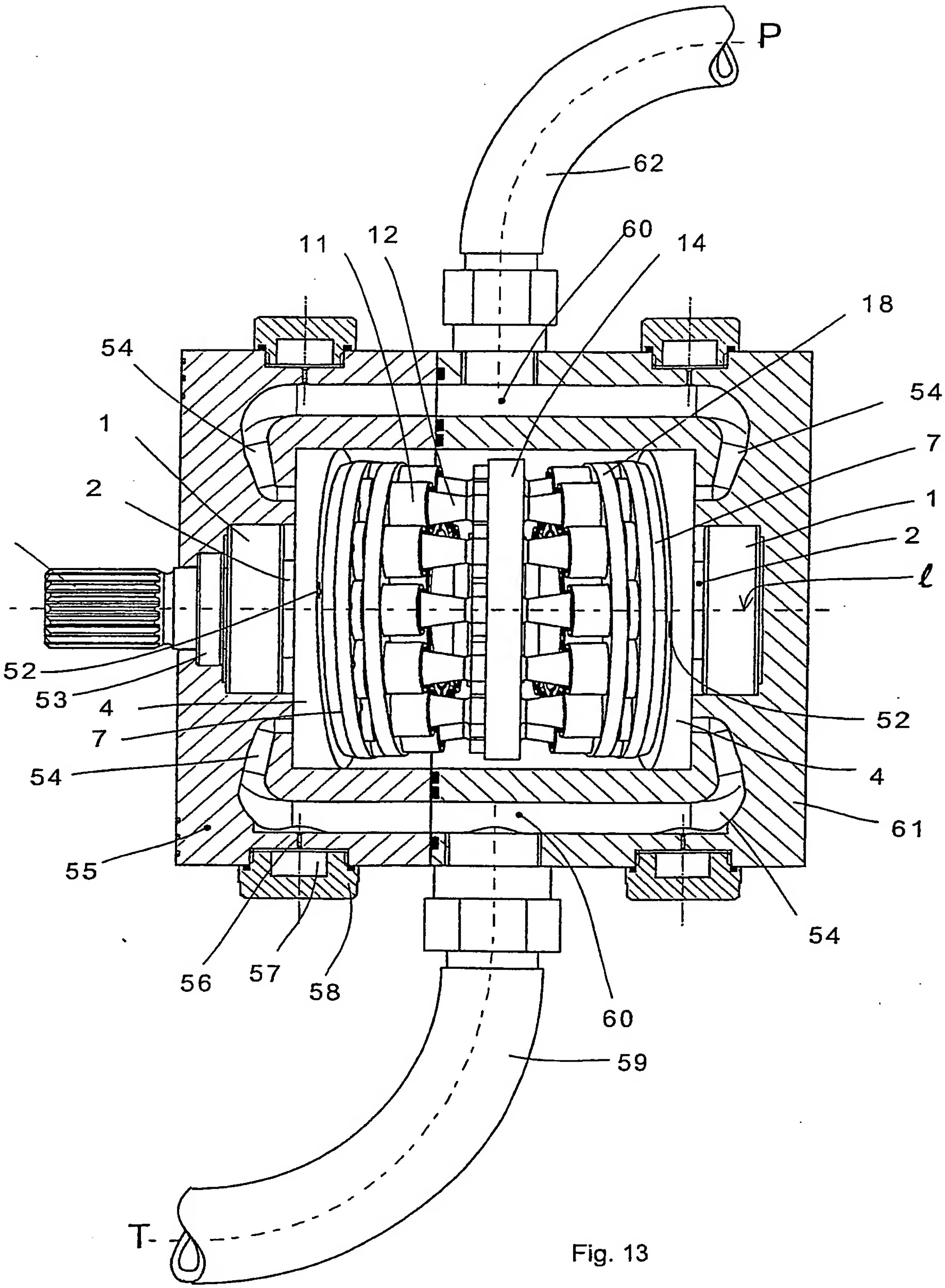
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Fig. 13

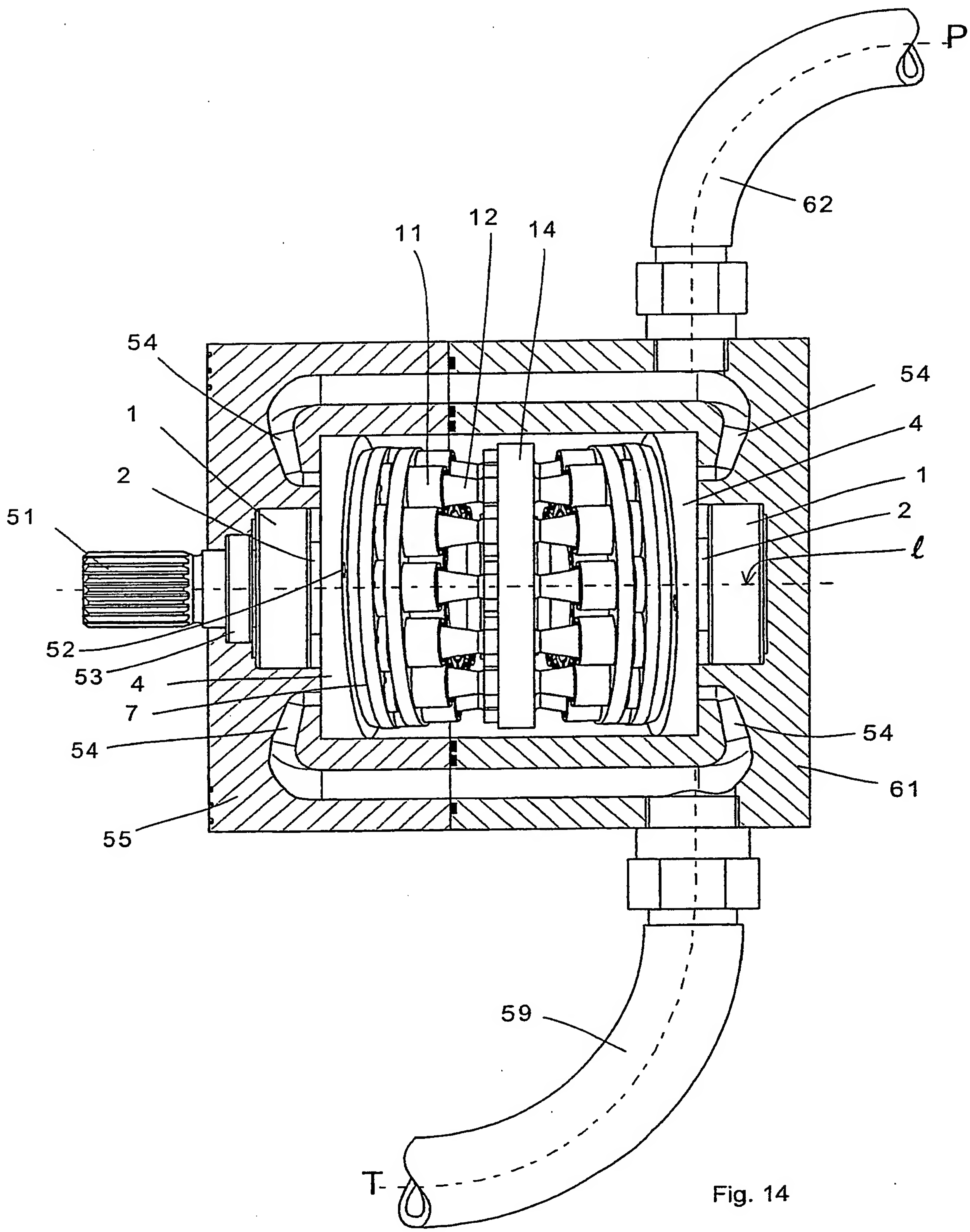

Fig. 14

INTERNATIONAL SEARCH REPORT

PCT/NL 03/00015

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 F01B3/00 F04B1/20

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 7 F01B F04B F03C

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal, WPI Data, PAJ

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	DE 35 19 783 A (DANFOSS AS) 4 December 1986 (1986-12-04) figures 1-3 abstract	1
A	US 3 434 429 A (GOODWIN RICHARD R) 25 March 1969 (1969-03-25) cited in the application figure 1 claim 1	1
A	DE 21 30 514 A (LINDE AG) 21 December 1972 (1972-12-21) figure 1 page 7, paragraph 1 -page 8, paragraph 1	1

-/--

[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

° Special categories of cited documents :

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"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
19 March 2003	25/03/2003

Name and mailing address of the ISA	Authorized officer
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C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication,where appropriate, of the relevant passages	Relevant to claim No.
A	US 3 648 567 A (CLARK WILLIAM B) 14 March 1972 (1972-03-14) figure 1 abstract column 1, line 1 - line 75 -----	1

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INTERNATIONAL SEARCH REPORT

Information on patent family members

PCT/NL 03/00015

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US 3648567	A	14-03-1972	NONE		